The Association

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Engineering and Shipbuilding Draughtsmen.

AN INTRODUCTION TO GAS TURBINES.

By WILLIAM L. GROSS, B.Sc. (Eng.), G.I.Mech.E.

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AN INTRODUCTION TO GAS TURBINES.

By WILLIAM L. GROSS, B.Sc. (Eng.), G.I.Mech.E.

I.—INTRODUCTION.

Gas turbines, and more so jet engines, have, during the last few years, captured the imagination of the public, but not much general information on the subject has yet been published. The object of this pamphlet is in the strictest meaning of the word to serve as an introduction and to stimulate interest in this fairly new type of power-generation.

With this aim in view, the writer has kept as far as possible to the descriptive side, and it is hoped that the mathematics and thermodynamics introduced will be easily understood by the patient reader. Wherever it seemed desirable, some basic definitions and conceptions have been added in order to facilitate an understanding of the subject.

II.—BRIEF OUTLINE OF HISTORY AND DEVELOPMENT OF GAS TURBINES.

Although having been an engineering proposition for a few years only, the gas turbine can look back on a long and interesting history, extending over hundreds of years. Hero of Alexandria in the year 130 B.C. is known to have experimented with turbines, using hot air as a driving medium, although it is problematical whether the term "gas turbine," as we understand it to-day, could be applied to his machine.

In the year 1791 an Englishman, John Barber, took out a patent which, embodies although in a very primitive way all the essential features of the modern counterpart, namely, gas and air compressors, a combustion chamber for gas and air mixture and a gas jet impinging on an impulse wheel.

Another Englishman, John Dumbell, in 1808, patented his version of a gas turbine, this being merely a rotor with blades placed in a stream of products of combustion.

Other patents included one by Bresson of Paris in 1837, and by Dr. F. Stolze of Charlottenburg in 1872, the latter one comparing more closely with modern turbines as it comprised an axial flow compressor. None of the above-mentioned designs was a power generating proposition owing to the dearth of knowledge which then existed about aerodynamics.

With the advent of steam turbines and especially since 1884, when Sir Charles Parsons took out his original patent, more time was spent on the problems of blade contours and angles, aerodynamics and improving the efficiency of turbines and axial compressors, the latter ones also being experimented with by the Parsons Company subsequent to the year 1900.

It can safely be said that the first systematic design of a gas turbine operating on a cycle as is being employed to-day was undertaken by Armengaud and Lemale at the Societé des Turbomoteurs in Paris during the period 1903-1906. This design embodied a multi-stage air compressor, a combustion chamber supplied with liquid fuel and an impulse turbine equipped with water cooling of blades and disc. The thermal efficiency obtained was just under three per cent., and although this may seem extremely low, it must be understood that this was the first time that any useful work had been produced in this novel way.

In 1908, Karavodine of Paris built a turbine operating on the explosion cycle (see later chapters) and in the same year Dr. H. Holzwarth began his work on explosion turbines. The result of his activity in that field has persisted till the present day, although the number of explosion turbines in existence is being eclipsed by the modern combustion turbine.

The earliest practical application of gas turbines was, perhaps, in the early 1930's in the shape of the Velox boiler. There the exhaust gases were used to drive a gas turbine coupled to a compressor, the latter one being used for supercharging in order to ensure better combustion.

Coming right up to modern times the Buchi supercharger may be mentioned here, this exhaust gas turbine having been used for many years with very satisfactory results. In 1936 a gas turbine, manufactured by the Brown-Boveri Company, was installed in the U.S.A. and used in the production of high octane gasolene. Especially during the last war, gas turbines in the form of jet engines have received an appreciable impetus in this country, mainly owing to the pioneer work in that field by Air-Commodore Sir Frank Whittle and a great amount of extremely useful data and information has been gained from the successful operation of these engines. Lastly, it may be said that there is in use to-day a gas turbine-electric locomotive, as well as a power station employing the same prime mover.

III.—THE CASE FOR THE GAS TURBINE.

The question quite naturally arises, why gas turbines ever came into the forefront and why so many engineers and designers spent so much time developing this prime mover at a period when the reciprocating steam engine was in universal use. The main

stimulus for the development was the quest for simplicity in design and operation. The motion of turbines is purely rotational so that there is no need to convert reciprocating motion into rotational motion as in steam engines, diesels and petrol engines. A source of power loss is thereby eliminated and so are stress reversals in the reciprocating parts and impulse loads on the bearings.

Furthermore, compared with all machines using steam as a working medium (reciprocating engines and steam turbines) the heavy and cumbersome boiler can be dispensed with. Connected with that is the fact that no water is required to fill the boiler and to supply the condenser. The operating medium for gas turbines, namely air, is universally available and independent of the location of the unit.

As for starting the machine, all that is required is literally "to press a button" and the set reaches its operating speed in anything from 5 to 15 minutes. The stoking up of the boiler hours beforehand is therefore a thing of the past. Also, as will be seen in later chapters, the number of auxiliaries is reduced.

Opposing the above-mentioned advantages it must be said that the thermal efficiency still leaves room for improvement. Present-day figures have gone up to approximately 30% with closed cycles, but the writer is convinced that this is not finality. This figure, however, compares favourably with the thermal efficiency of 8-13% of the reciprocating steam locomotive.

One of the most important problems which retarded the advent of gas turbines for a considerable time was the necessity for materials to withstand elevated temperatures of the order of 700-800°C., this being used for the turbine blades. Advances in metallurgy are, therefore, of paramount importance in the development of the modern turbine.

IV.—THE BASIC GAS TURBINE SET.

A diagrammatic sketch of the principal components of a modern internal combustion gas turbine is shown in Fig. 1. This set is depicted driving an electric generator, but can naturally be used as prime mover for almost any type of machinery.

Atmospheric air is compressed in an axial flow compressor (or any other type of compressor) and then passed into the combustion chamber as shown. In that chamber, into which is injected gaseous or liquid fuel, some of the air is mixed with the fuel and burned with it, the other passes through an annular space (not shown) and then mixes with the products of combustion and cools these to a suitable turbine inlet temperature of the order of 500-850°C. This high temperature gas is then expanded in the turbine, thus doing useful work on the turbine rotor.

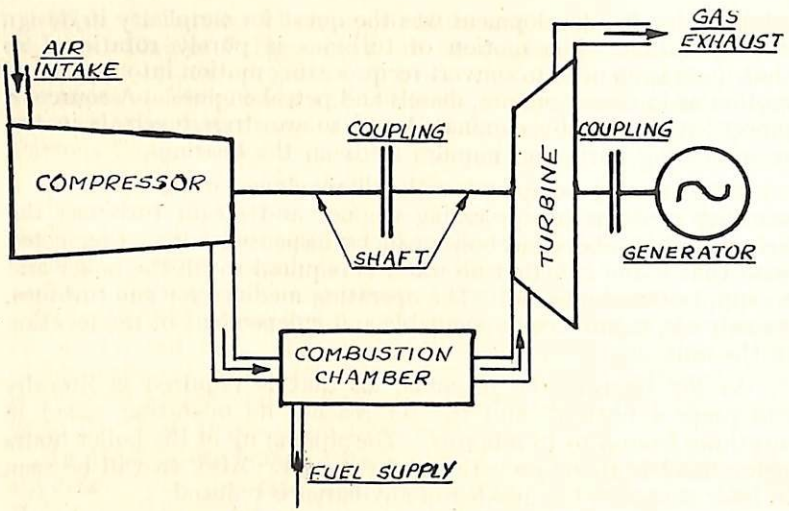

Fig. 1.—Diagram of Simple Gas Turbine Set.

In Fig. 1 the turbine proper is coupled to the compressor as well as to the generator, so that all the power developed by it is consumed by these two units. Only a part of the turbine output is therefore available for external use and it can now be seen why so many early attempts at designing a unit did not give the required results, the power developed by the turbine in those days being only just sufficient to drive the compressor or at the best provide only a small amount of surplus power.

This point of power distribution in a gas turbine set may best be illustrated by actual figures. In a representative set of say 2500 H.P. output, the turbine itself may develop something like 7500 H.P., of which the difference, namely 5000 H.P. is utilized for driving the compressor. These figures naturally vary with turbine inlet temperature and the ratio of useful power to total power generated increases with it. Nevertheless this fact of large power absorption by the compressor remains and must be constantly borne in mind by gas turbine designers.

Starting the set is effected by means of an electric (or some other) starting motor (not shown), which is coupled to the compressor-turbine shaft possibly in conjunction with a throw-out mechanism which comes into operation once the required speed has been reached. This is to prevent the motor being driven by the set. After starting, the fuel in the combustion chamber is ignited and once the self-driving speed is reached the set accelerates up to its operating speed. A speed governor, regulating the fuel supply and thereby controlling the speed, is one of the auxiliaries required.

As for the functioning of the turbine itself, the reader is referred to the large quantity of literature now available on steam turbines, but some of the more salient points of difference between gas and steam turbines may be mentioned here. Gas turbines usually have fewer stages than steam turbines of similar output, one stage being a row of rotor and stator blades. The diagram of a turbine is shown in Fig. 2.

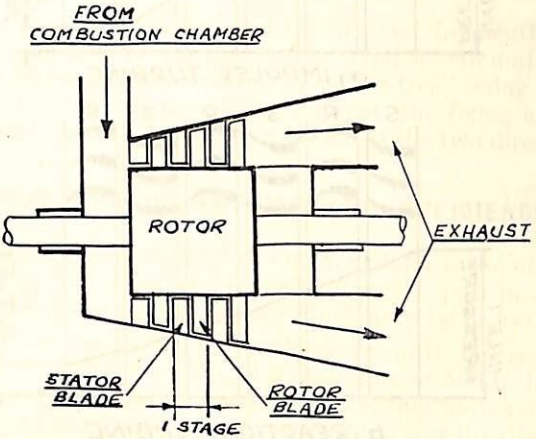

Fig. 2.—Diagram of Three-Stage Gas Turbine.

Reaction turbines are almost universally employed to-day, whereas with steam turbines impulse turbines are frequently encountered. The difference between the two may best be explained by means of a sketch, Fig. 3. In impulse turbines, the working medium is expanded in the fixed blades (stators) only, and the pressure remains constant over the moving blades (rotors).

In reaction turbines there is no sudden pressure drop, but the drop is gradual and takes place continuously as the medium passes over rotors and stators. Fig. 3 shows a section through the blades and depicts the variation of pressure as the working medium traverses the stages.

Gas turbines usually work with very large velocities and therefore a large throughput. This naturally means heavy gas-bending stresses which, coupled with high centrifugal blade stresses and elevated temperatures result in the blade material having to work in extremely unfavourable conditions and explains the fact that blades must be made of the best quality material obtainable, material which has only been developed within the last few years.

Blade cooling both for stators and rotors has been investigated especially in German gas turbines. German engineers did not have

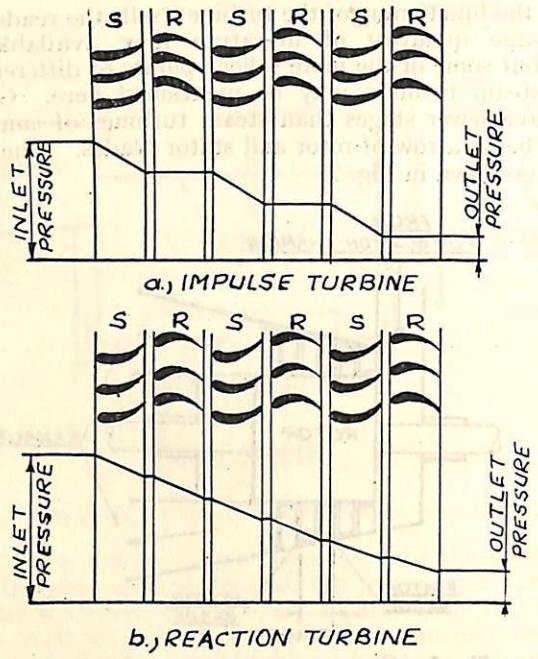

Fig. 3.—Pressure Variation in Impulse and Reaction Turbines.

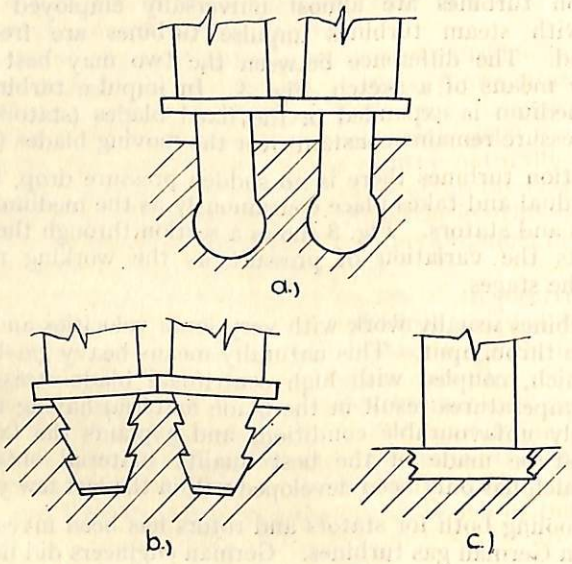

Fig. 4.—Systems of Blade Fixing.

at their disposal the high-grade material required and therefore had recourse to hollow air or water-cooled blades.

The means of fixing the blades to the rotor may also be of interest, particularly as this led to frequent trouble in the early days of gas turbines. Blades were apt to work loose under the high centrifugal forces set up and often became detached, thereby wrecking both rotor and stator casing. Various systems of blade fixing are shown in Fig. 4.

Blade roots, as shown in (a) and (b) are met frequently, slots being axial in the case of (a) and either axial or circumferential with (b). The latter is often referred to as "fir-tree" fixing. With short blades not subjected to excessive forces, the fixing as at (c) may be employed, again with grooves in one of the two directions.

V.—THERMAL CONSIDERATIONS AND EFFICIENCIES.

With any prime mover, one of the main criteria is the efficiency *i.e.*, how much of the amount of work (or heat) put in can be "got out" as useful work. Some definitions may be of use here.

Work is measured in ft. lb. units and power in ft. lbs./sec. units. One horse-power is defined as 33,000 ft. lbs./min. or 550 ft. lbs./sec., *i.e.*, power is the rate of doing work. On the thermodynamic side one of the units we shall most frequently encounter is the Centigrade heat unit, more commonly referred to as C.H.U. and is defined as the amount of heat required to raise 1 lb. of water through 1°C. If, as always in heat engines, heat energy is converted into mechanical energy, we must know the relationship between C.H.U. and ft. lb. units. This is—

$$1 \text{ C.H.U. heat} = 1400 \text{ ft. lbs. work,}$$

and we can now define thermal efficiency as follows :—

$$\text{Thermal efficiency} = \frac{\text{Output}}{\text{Input}} \times 100\%$$

$$= \frac{\text{Output ft. lbs. work}}{\text{Input C.H.U. heat} \times 1400} \times 100\%$$

In the case of gas turbines, heat is put in by the fuel injected into the combustion chamber.

Considering the working of the compressor and turbine separately we must also consider their respective efficiencies. As will be readily understood, neither of these two units as such are of 100% efficiency owing to losses caused by friction, windage, gas and air leakage, duct losses and such like. If they were 100% efficient the set would be self-driving, without adding heat in the form of fuel, but they would not deliver any useful work.

Present-day designs have an overall efficiency (turbine × compressor efficiency) of 70-75% and Fig. 5 shows the effect of this on the overall thermal efficiency of an ideal set, as Fig. 1, with turbine inlet temperature of 800°C., as well as on the work ratio (nett output of set over output of turbine proper). From these graphs it is evident that the increase of the overall efficiency has a marked effect on the thermal efficiency and also illustrates the reason why early gas turbines with an overall efficiency of about 50% (turbine 78%, compressor 68%) could not work.

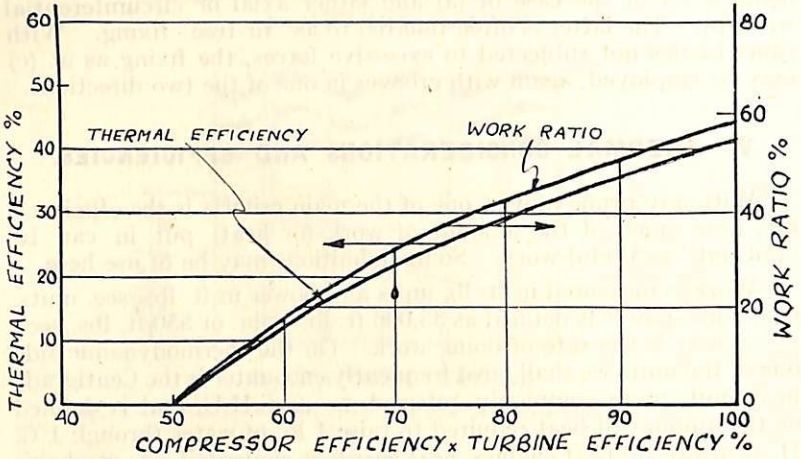

Fig. 5.*—Efficiencies of Ideal Cycle Gas Turbine Set. (Inlet temperature = 800°C.).

As more work is done in the turbine than is absorbed in the compressor, the benefit from a given improvement in turbine efficiency is greater than from the same percentage improvement in compressor efficiency, a fact which is possibly not evident from Fig. 5, but one well recognised by designers.

Two more variants have a considerable bearing on the performance of a gas turbine set, namely the compressor pressure ratio and the turbine inlet temperature, the former a problem for the designer and the latter for designer and metallurgist. Fig. 6 shows these variants and the thermal efficiency on one graph.

A Ideal cycle without duct losses, turbine × compressor η = 100%
B ,, ,, ,, ,, ,, η = 85%
C Actual cycle, turbine inlet temperature = 810°C.
D Actual cycle, turbine inlet temperature = 560°C.

* From Dr. Meyer : History and Development of Combustion Gas Turbine.

A comparison of curves A and B shows the effect of turbine and compressor efficiencies on the thermal efficiency of the whole set whilst curves C and D illustrate the reduction of thermal efficiency with turbine inlet temperature. The problems of compression ratio and inlet temperature are also intimately connected and must be considered with regard to each other as elevated temperature is desirable in order to raise the thermal efficiency. In order to operate at the peak point an increase in compressor pressure ratio is also necessary.

The question of turbine inlet temperature is more clearly shown in Fig. 7, also depicting the fact that until materials were evolved to withstand an operating temperature of about 360°C., no gas turbine could be a working proposition.

As is to be expected, the ambient (locally prevailing) temperature of air has a marked effect on the output, and gas turbines have to be designed according to the climatic conditions prevailing at their location of operation. This can be seen from Fig. 8.

In a pamphlet of this size the foregoing must suffice as regards the question of efficiencies of a gas turbine set and how they are affected by other variants. But no description of any heat engine would be complete without discussing the thermal cycle.

In order to do this, we need some definitions which are given below.

Total Heat of a gas is the sum of its internal energy (*i.e.*, the heat energy stored in the gas) and its pressure energy. Total heat is measured in CHU/lb. and difference in total heat of a gas before and after expansion gives a measure of the work done by or on it.

Entropy of a gas is best defined as a function of its state. The total amount of entropy possessed by a gas cannot be measured, but the change of entropy can. If Φ (Greek letter phi) be the symbol for entropy, the change is given by

$$\delta \Phi = \frac{\delta H}{T}$$

where $\delta \Phi$, and δ H mean change in entropy and total heat respectively and T is the absolute temperature at which this change takes place.

Isothermal expansion or compression of a gas takes place if the temperature of the gas remains constant and the change follows the law $PV = \text{constant}$.

Adiabatic expansion or compression of a gas takes place without heat being added to or taken from it. Expansion follows the law

$$PV^{\gamma} = \text{constant} \quad (\gamma = 1{\cdot}4 \text{ for air})$$

where P is absolute pressure and V is unit volume.

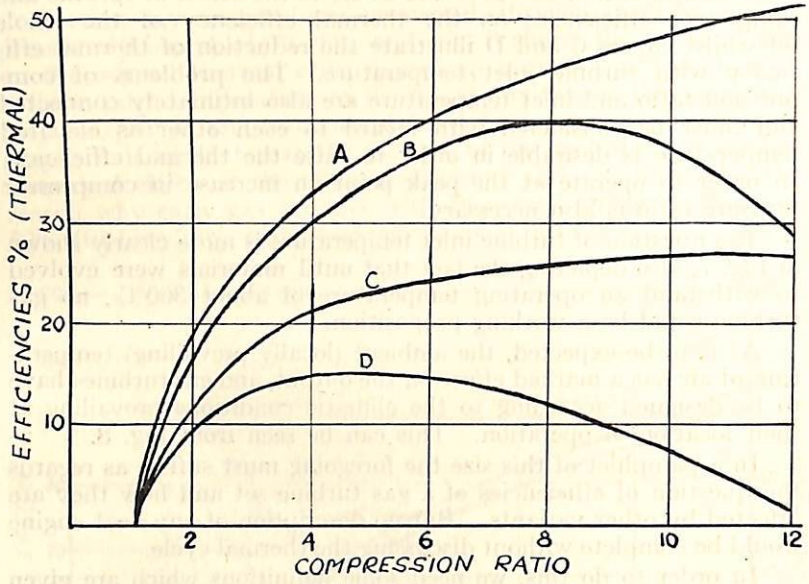

Fig. 6.*—Efficiencies, Inlet Temperature and Compression Ratio of Simple Cycle.

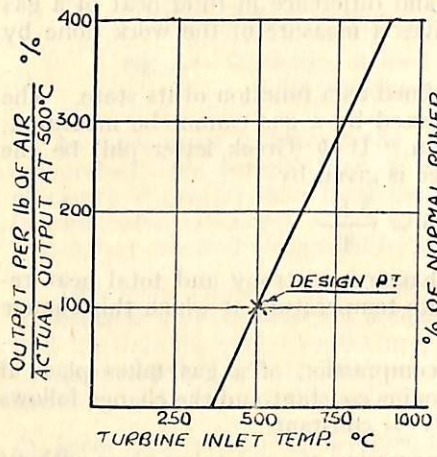

Fig. 7.†—Effect of Turbine Inlet Temperature.

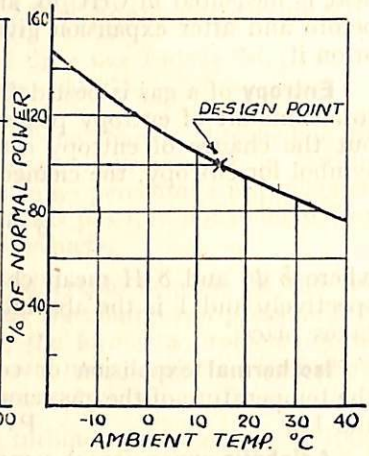

Fig. 8.†—Effect of Ambient Temperature.

* Curves A, C and D from Dr. J. T. Rettaliata, "The Gas Turbine."
† Figs. 7 and 8 from Dr. J. T. Rettaliata, "The Gas Turbine."

The most common way of depicting a heat engine cycle is on the pressure-volume diagram, *i.e.* a graph showing the variation of the pressure and volume of 1 lb. of gas as it traverses the engine. Considering the diagram, Fig. 9, line AB represents the (adiabatic) compression of the gas in the compressor from the outside air pressure P_2 to the combustion pressure P_1 and from a volume V_1 to V_2. In the combustion chamber the air burns with the fuel, thereby expanding along BC from V_2 to V_3 at constant pressure P_1. In the turbine proper the gas expands (adiabatically) along CD to V_4 and down to the outside air pressure P_2. The cycle is completed by the removal of heat from the exhaust gases in atmosphere along DA.

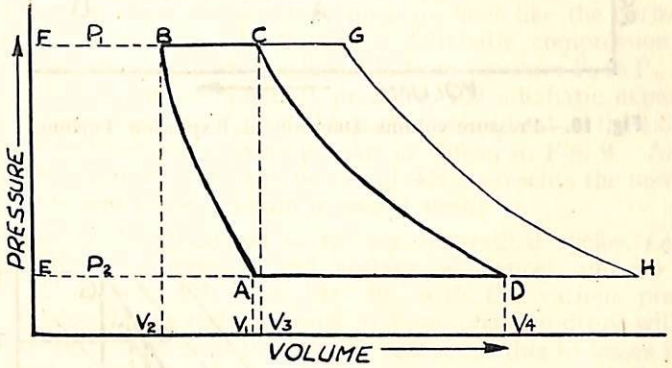

Fig. 9.—Pressure-volume Diagram of Gas Turbine.

In any PV diagram the enclosed area represents the work done on or by the gas. Accordingly, the area FBAE represents the work to be done by the compressor per lb. of air, whereas area FCDE represents the work output of the turbine per lb. of air. Therefore the excess area ABCD represents the work available for external use per lb. of air. It must be understood however that this is a purely theoretical cycle and does not take into account compressor and turbine efficiencies or pressure losses in the cycle.

In the first chapter, reference was made to the explosion gas turbine and the working of that will be explained with the aid of a PV diagram, Fig. 10. Line AB represents pre-compression (by means not specified) of the charge. Explosion (*i.e.*, rapid combustion) at constant volume is along BC, there being no time for the gas to expand as previously, and a sudden rise in temperature ensues. Expansion in the turbine CD and removal of heat from the exhaust gases DA are as in Fig. 9. The paramount difference between the two types of gas turbines lies therefore in the combustion process, one being at constant pressure and the

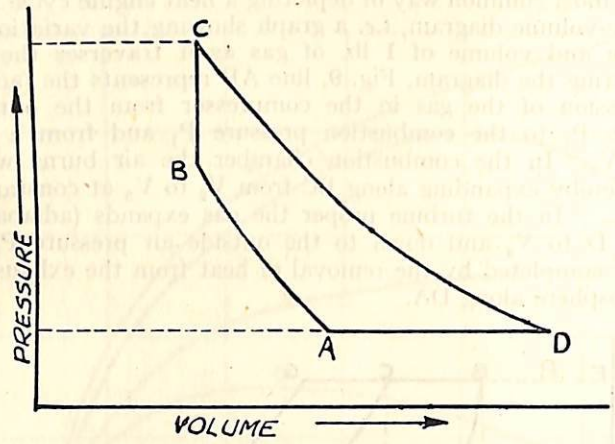

Fig. 10.—Pressure-volume Diagram of Explosion Turbine.

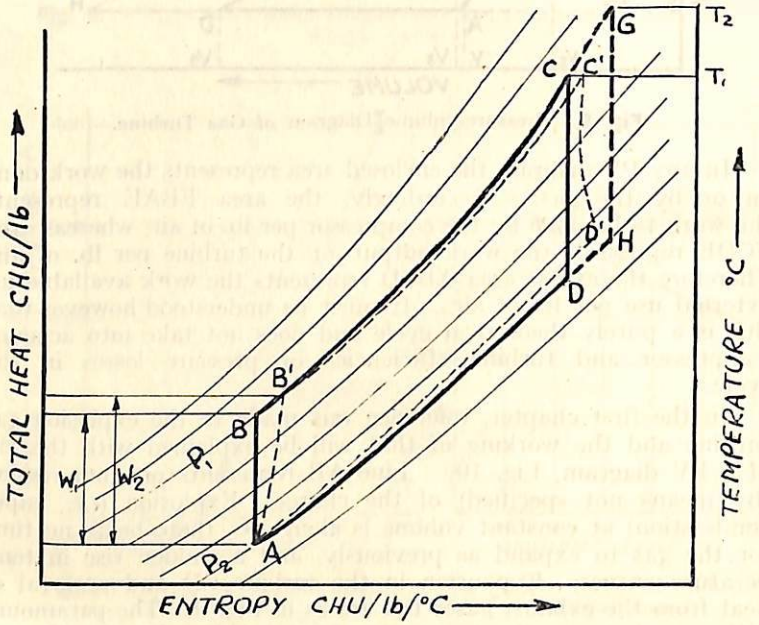

Fig. 11.—Entropy-total heat/temperature diagram of Ideal Cycle.

other one (explosion) at constant volume. As the explosion turbine is not met very frequently to-day, all further references to gas turbines in this pamphlet will therefore be understood to mean combustion turbines.

Referring back to Fig. 9, the thin line GH shows the diagram if the turbine inlet temperature were increased, resulting in an increase in useful work by CGHD, with the same amount of work to be done by the compressor FBAE.

Another method of depicting a heat engine cycle is on a temperature-entropy and total heat-entropy diagram, as on Fig. 11. In this type of diagram pressure, temperature, entropy and total heat at any stated point can be read off directly. The curves traversing the chart are constant pressure lines like the horizontal pressure lines on the PV diagram. Adiabatic compression (*i.e.* constant entropy) follows the line AB from pressure P_2 to P_1. BC depicts combustion at constant pressure, CD adiabatic expansion in the turbine down to P_2 and DA is the loss of heat of the exhaust gases, all this being the same process as shown in Fig. 9. Also, if the turbine inlet temperature be raised GH represents the new part of the cycle and there is a gain in useful work.

All the cycles discussed so far are theoretical cycles, *i.e.*, assuming 100% compressor and turbine efficiences and no duct losses. The cycle ABCD in Fig. 11, with the various pressure drops, is also shown and the cause of these pressure drops will now be explained. The drop between B′ and C′ is due to losses in the duct between compressor outlet and turbine inlet and the drop between D′ and A occurs in the exhaust duct from the turbine. Deviation of the line AB to AB′ and similarly from CD to C′D′ is caused by a phenomenon known as "reheating" and is inherent in all turbines and rotary blade compressors. As the gas passes over the blades, friction reduces its velocity, increases its temperature and thereby the total heat from B to B′. As the change in total heat gives a measure of the work put in or obtained, it results in W_2 (*i.e.* more than W_1) being put in by the compressor for the same pressure ratio and also less output being obtained from the turbine. This results in an overall drop in efficiency.

The mathematical expression for excess energy per lb. of air for an ideal cycle without losses, is given by

$$h = \frac{144\ \gamma}{1400\ (\gamma - 1)} P_1\ (V_c - V_b) \left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right]$$

where the suffixes refer to the points on the diagram, Fig. 11, and γ is the index of the compression and expansion curves AB and CD respectively, following the law

$$PV^{\gamma} = \text{constant.}$$

The above equation ignores the addition to the mass flow due to the fuel added.

VI.—HEAT EXCHANGERS.

As in so many other heat engines, heat exchangers are extensively used with gas turbines, but their use must be judicial or the result desired will not be achieved.

If one realises that the temperature of the exhaust gases from the turbine can be of the order of 500°C. for industrial sets and as high as 650°C. for aircraft gas turbines, it will be readily understood that a heat exchanger can materially affect the thermal efficiency of a gas turbine set, besides reducing the exhaust temperature to a more reasonable figure. A revised diagram is shown in Fig. 12.

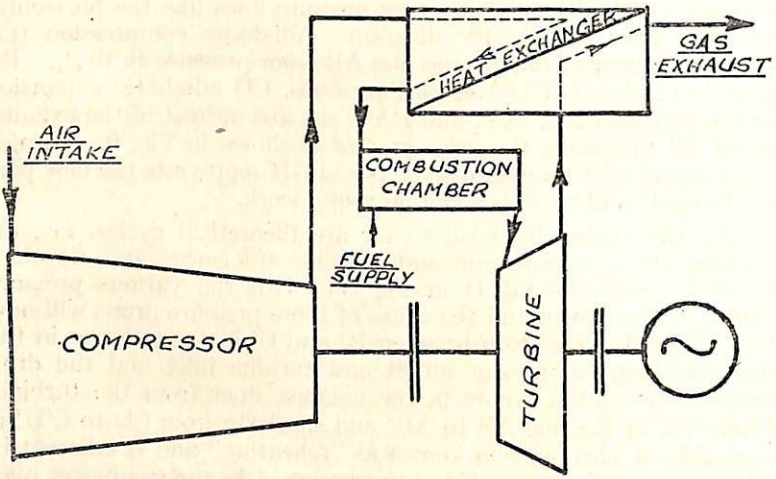

Fig. 12.—Gas Turbine Set with Heat Exchanger.

This unit is installed in the set between compressor and combustion chamber on the air side and between turbine and exhaust on the gas side. The compressed air is heated thereby before fuel injection, so that the amount of fuel which has to be burned for the gas to reach the same temperature is reduced. As the amount of fuel gives a measure of the heat put in, it will be readily understood that the thermal efficiency is thereby increased.

The mechanical construction of heat exchangers, is of the tubular or plate type and they are usually very heavy and cumbersome pieces of apparatus.

In the tubular type, tubes through which the exhaust gases pass are placed in the path of the compressed air or *vice versa*, and heat is exchanged through the tube walls. With the plate construction there are alternate layers of gas and air between the plates. As for the classification on the thermal side, there are

counterflow, parallel flow and crossflow exchangers, each having its own characteristics.

According to Fig. 13, air increases in temperature from Ta_1 to Ta_2, whereas the gas cools from Tg_1 to Tg_2. In the counterflow type the temperature difference between the two fluids remains practically constant and so does the rate of heat transfer, which is proportional to the temperature difference.

With the parallel flow type, the temperature difference decreases continuously and the overall rate of heat transfer is less favourable than in case (*a*).

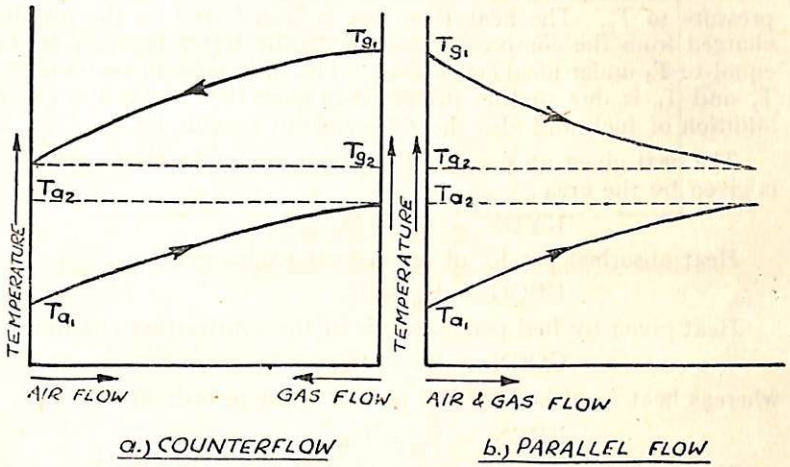

Fig. 13.—Types of Heat Exchangers.

As regards crossflow exchangers, the temperature distribution is more complex and can only be represented in a three-dimensional diagram. One point worth noting with parallel flow heat exchangers is the fact that if they were 100% efficient, Tg_2 and Ta_2 would be identical, a state which is never realised in practice.

A lightweight heat exchanger of novel design has been developed by a German engineer, Dr. Ritz of A.V.A., Gottingen. It is of the rotary type and consists of a ceramic cylinder in two parts, through which the hot and cool gases pass alternately in a radial direction. The walls are alternately heated and cooled by the gases and heat transfer is achieved *via* the walls. Rotational speed is adjusted to allow the walls to reach their correct temperatures. Such a unit designed for an aircraft gas turbine developing 5000 shaft H.P. at 10,000 meters altitude weighed only 1650 lbs. A tubular or plate type exchanger for a plant of similar output, would weigh approximately 4 tons.

There are, therefore, two distinct types of heat exchangers classified by the method of heat transfer, namely recuperative and regenerative. In the former, heat is passed directly from gas to air *via* the dividing wall, whereas in the latter (as with the rotary exchanger) the heat from the gas is stored in the walls and then given up to the air after each change of flow.

The role of the heat exchanger in the thermodynamic cycle is best explained by reference to Fig. 14. This depicts the ideal gas turbine cycle ABCD, as described in Chapter V. If we assume a counterflow heat exchanger of infinite surface, *i.e.* 100% efficiency, the turbine exhaust gases will be cooled down from T_d at constant pressure to T_y. The heat thus lost is transferred to the air discharged from the compressor and heats the latter from T_b to T_q, equal to T_d under ideal conditions. The difference in temperature T_y and T_b is due to the difference in mass flow of gas and air (in addition of fuel) and also the difference in specific heats.

The heat given up per lb. of exhaust gas in the heat exchanger is given by the area

$$KYDN = H_d - H_y$$

Heat absorbed per lb. of air in heat exchanger

$$EBQG = H_q - H_b$$

Heat given by fuel per lb. of air in the combustion chamber,

$$GQCN = H_c - H_q,$$

whereas heat furnished by fuel per lb. of air in ordinary cycle,

$$EBCN = H_c - H_b.$$

Therefore the thermal efficiency of a heat exchanger cycle is by

$$\left[\frac{H_c - H_b + K^1}{H_c - H_q + K} - 1\right] \times 100\%$$

greater than the corresponding efficiency of a simple cycle, each time neglecting pressure and other losses. K is the amount of heat added by the fuel per lb. of air for the heat exchanger cycle and K^1 the corresponding amount for the simple cycle.

Our next consideration is the actual cycle, with an infinite surface heat exchanger, in Fig. 15. Again using the previous notation, ABCD represents the ideal cycle and AB'C'D' the actual cycle incorporating the various losses. The heat given up per lb. of gas in the heat exchanger is given by the area LUD'W, which is less than the difference $H_b - H_y$ in the previous case, as the line D'U is not a constant pressure line (a fact which is proved in any text-book on thermodynamics). Similarly, the area FB'SJ, the heat absorbed per lb. of air, is less than $H_q - H_b$ previously. Using the same reasoning as before, T_u and T_b' will not coincide, although the difference will be less than $T_y - T_b$ in Fig. 14 due to the pressure

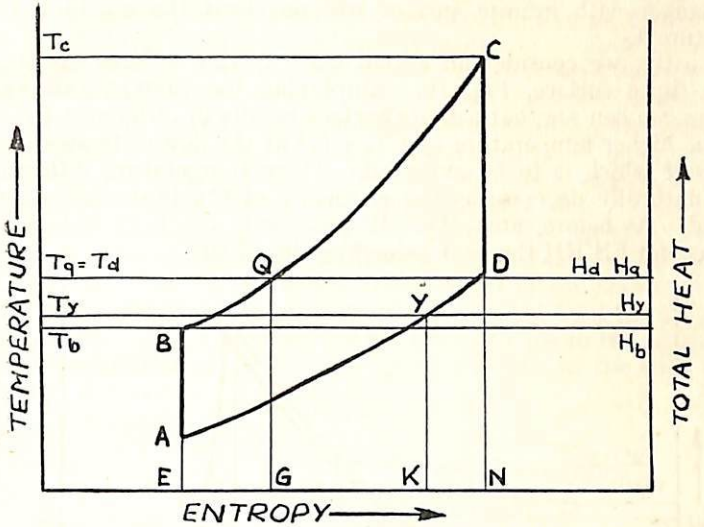

Fig. 14.—Ideal Cycle with Heat Exchanger of Infinite Surface.

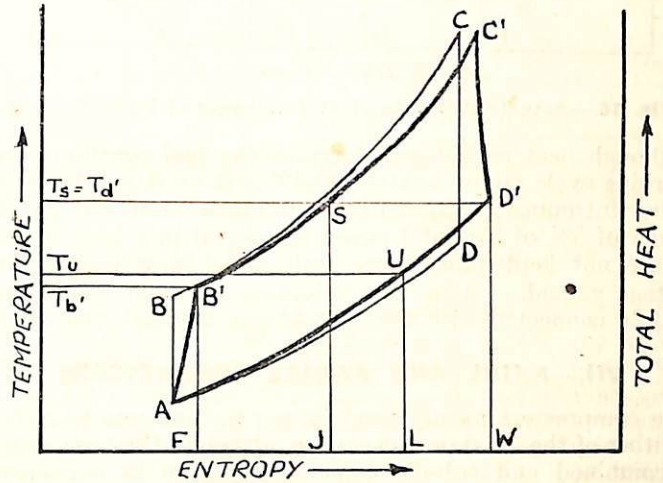

Fig. 15.—Actual Cycle with Infinite Surface Heat Exchanger.

losses in the actual cycle. If these become excessive, even a heat exchanger with infinite surface will not heat the air to a temperature T_d'.

Lastly, we consider an actual cycle having a heat exchanger with finite surface, Fig. 16. Employing the same notations as before, we can see that a finite surface results in difference $T_d' - T_r$ at the higher temperature and $T_v - T_b'$ at the lower temperature, a result which is to be expected. These temperature differences will naturally decrease as the efficiency of the heat exchanger is raised. As before, area MVD'W represents the heat lost per lb. of gas and FB'RH the heat gained per lb. of air.

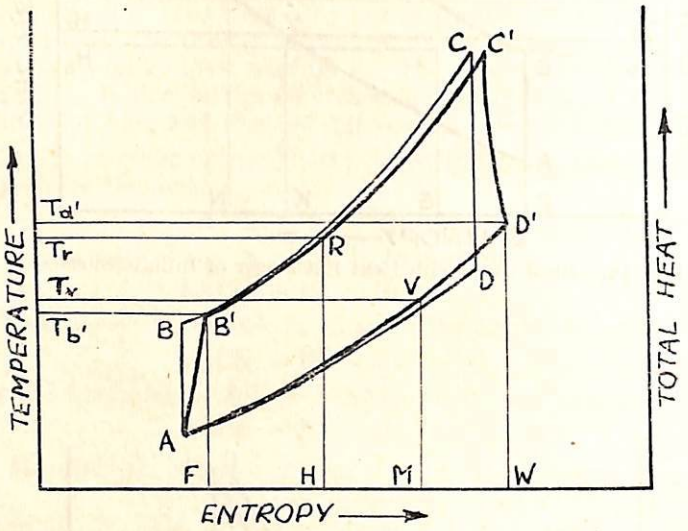

Fig. 16.—Actual Cycle with Heat Exchanger of Finite Surface.

Although heat exchangers decrease the fuel consumption of a gas turbine cycle approximately 30-40%, it must not be forgotten that they introduce additional pressure losses on air and gas side of the order of 5% of the total power developed in a large set, and if these are not kept down may well affect any thermodynamic advantage gained. Also, the adoption of heat exchangers is intimately connected with the type of gas turbine cycle chosen.

VII.—AXIAL AND RADIAL COMPRESSORS.

The compressors mostly used in gas turbine sets to date have been either of the axial or radial type, although they are also being used combined and sub-divisions of each are in existence. In considering a gas turbine set therefore, it is advantageous to know the relative merits and characteristics of each.

The functioning of the axial compressor can be looked at as a turbine in reverse, since it consists essentially of rotor drums and blades and stator casings and blades as in Fig. 17. Air is drawn in axially, as shown, but in passing through the first rotor stage acquires in addition to its axial velocity a tangential component, the resultant velocity being greater than the original axial velocity. This is evident from Fig. 17. The stator redirects the air into a direction more or less axial, depending on design conditions and the type of blading chosen, accompanied by a drop in velocity. The same process is repeated at every stage. The air thus undergoes an increase and decrease in kinetic energy (*i.e.* velocity energy) per stage, but the energy which is destroyed in the stators is not lost, but is converted into pressure energy as seen from the pressure diagram, Fig. 17. The crossflow section available to the air tapers towards the outlet so as to allow equal velocities at the ends with increased density.

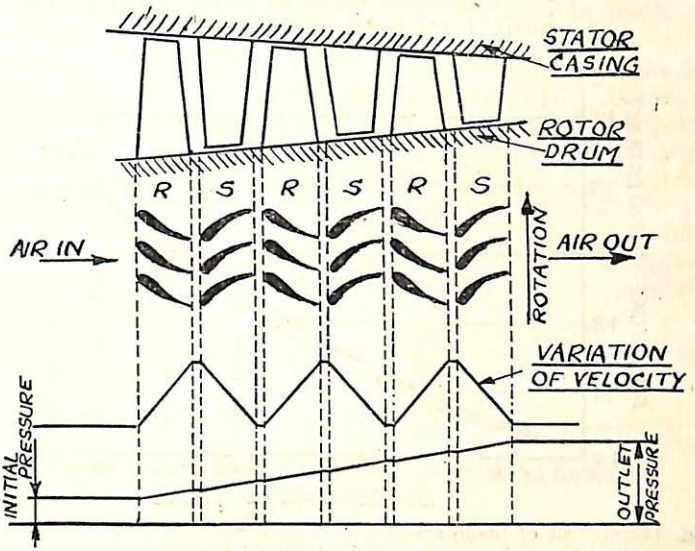

Fig. 17.—Diagram of Axial Compressor.

The development of the power formula for the ideal no-loss condition is beyond the limited space and elementary nature of this pamphlet and the reader is referred to the bibliography at the end. But since important conclusions on the performance of axial compressors can be drawn from it, it is quoted below.

Power input to compressor

$$= \frac{W\ u^2\ (1 - \tan \beta_1 \cot \beta_2)}{550\ g}\ \text{HP}$$

where W = mass flow of air lbs./sec.
u = circumferential speed of blades ft./sec.
g = 32·18 ft./sec.^{2}
β_1 = rotor blade inlet angle.
β_2 = rotor blade outlet angle.

The formula shows that power input is proportional to the square of the blade speed.

Again quoting the ideal stage pressure ratio formula for better understanding of the compressor,

$$\frac{P_2}{P_1} = \left[\frac{(n-1)\,u^2\,(1-\tan\beta_1 \cot\beta_2)}{g\,\gamma\,P_1\,V} - 1\right]^{\frac{n}{n-1}}$$

where P = absolute pressure.
V = specific volume.
n = index of compression.
γ = adiabatic index of compression (=1·4).
suffix 1 refers to inlet conditions.
suffix 2 refers to outlet conditions.

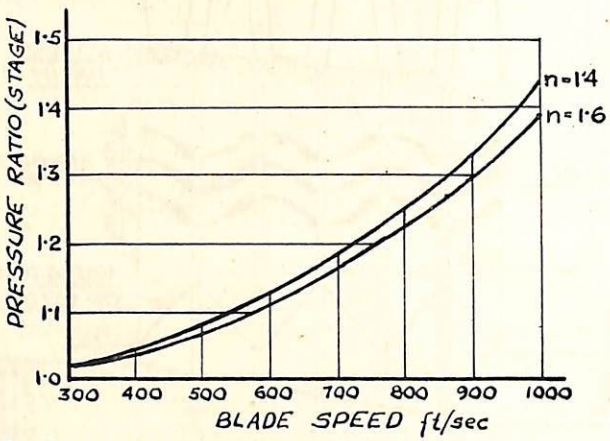

Fig. 18.—Effect of blade speed on pressure ratio (axial compressor). Inlet angle 30°, Outlet angle 40°.

This formula was used to plot approximately the graph shown in Fig. 18. It shows an increase of stage pressure ratio with blade speed, but not increasing in the square ratio as is the case with power input. Owing to the relatively low stage pressure ratio, axial compressors are always multi-stage.

In all machines the conception of efficiency must be introduced. In this case we must also differentiate between adiabatic and mechanical efficiency. The former is the ratio between ideal work required with adiabatic compression to that actually required.

This efficiency depends on blade form and stage pressure ratio. Mechanical efficiency on the other hand is the ratio of work done on air to work input and depends on bearing friction, windage and other mechanical considerations. The overall efficiency of the compressor is the product of the two.

Having mentioned the more important aspects of the internal characteristics, let us now consider the running characteristics and behaviour at load and part-load of an axial compressor. Clearly, the most important aspects are its massflow of air and the overall pressure ratio achieved.

Fig. 19 shows the relation between massflow and pressure ratio with different speeds, all compared to their design values. This shows that the massflow has a certain value for maximum pressure ratio at a given speed. Maximum efficiency also occurs near the maximum pressure ratio point. It will be seen later that a change in any of these three quantities, as happens in part-load working, results in decreased efficiency—a most undesirable result with compressors.

The next aspect to be discussed is surging, a phenomenon common with all axial and radial compressors. Assume the compressor to run at the design speed, Fig. 19, but at a point P to the right of the peak. This corresponds to a certain delivery pressure and massflow. If a valve be put into the compressor outlet and the massflow gradually decreased, the operating point will move along the curve towards the left until the maximum is reached. Any further closing of the valve will cause the operating point to move to the left of the maximum and the compressor

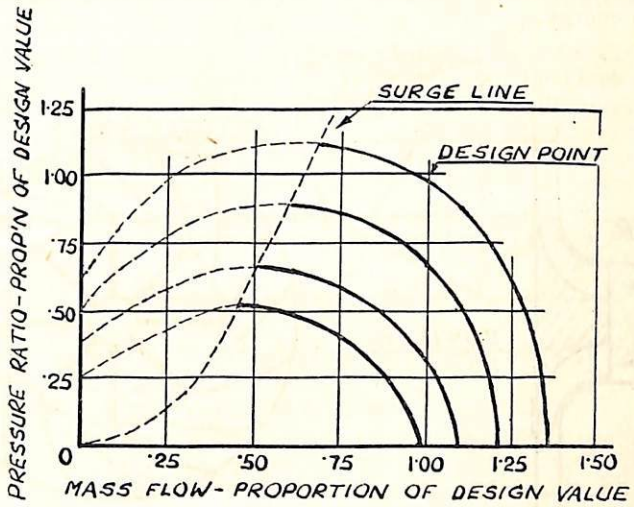

Fig. 19.—Pressure Ratio and Massflow of Axial Compressor.

delivery pressure will drop. But the pressure in the delivery system is not instantaneously affected by this drop and will momentarily be greater than the delivery pressure. There will then be a tendency for the direction of flow to be reversed, delivery will stop and the operating point drop to N. Air in the delivery system rapidly dissipates, causing a corresponding drop of pressure below the value at N, whereupon delivery will start again, the operating point move *via* P to its peak and the process will be repeated. Clearly this leads to unstable and pulsating flow and is not permissible.

It is probable that this explanation of surging does not tell the complete story and it is likely that owing to part-load working of the compressor the gas incidence angle does not agree with the blade inlet angle, the effect of this being cumulative with resulting general breakaway and turbulence.

Nevertheless it follows from this that any operation to the left of the peak points is unstable and the dotted line joining these is known as "surge line." Compressors are, therefore, designed to work to the right of this line as shown by Fig. 19.

Radial or centrifugal compressors are, unlike the type just discussed, frequently used in single stages and due to their simplicity are reliable. Their basic construction is shown in Fig. 20. A radial compressor consists essentially of a wheel, known as the impeller, with a number of blades on one or both faces, thus dividing the space between the wheel and casing into radial passages. The casing is provided with vanes curved in the direction of rotation leading into the outlet. This vane space is known as the diffuser and converts part of the kinetic energy possessed by the air into pressure energy.

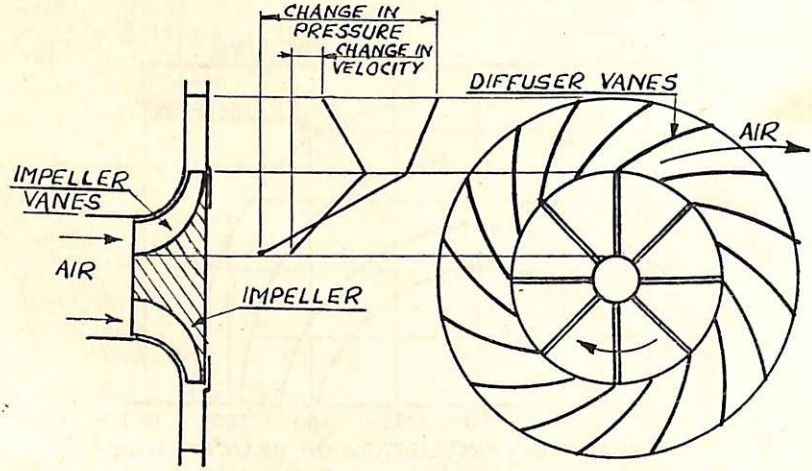

Fig. 20.—Diagram of Radial Compressor.

When the impeller starts rotating at high speed, the air contained in the radial spaces between the impeller vanes will rotate with it, and because of the centrifugal force will be forced outwards. The vacuum thus formed at the hub will cause more air to flow axially into the compressor and radially out towards the periphery, as shown. When the air leaves the impeller it will, therefore, have acquired both a radial and tangential component of velocity and the inlet angles of the diffuser vanes are designed to ensure shock-proof entry. As the object of the diffuser is to convert kinetic to pressure energy, there will be an increase in pressure and decrease in velocity as indicated.

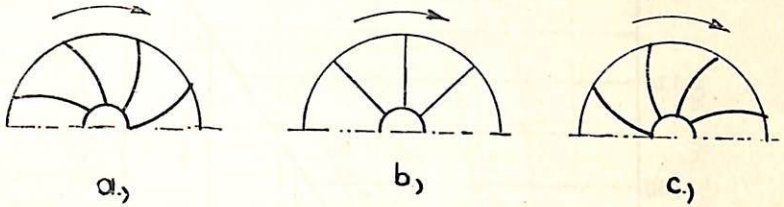

Fig. 21.—Shape of Impeller Vanes.

Three types of impeller vanes are shown in Fig. 21, each having its own characteristic. With forward leaning blades (*c*), the largest possible compression ratio is theoretically obtainable. It also imparts the largest absolute velocity to the air so that with this type the largest amount of kinetic energy has to be converted to pressure energy, calling for very careful design of diffuser. As diffuser losses, however, are very high, forward swept vanes are hardly ever used and their adoption depends on the improvement of diffusers yet to be achieved.

Backward swept vanes, as shown at (*a*), result in the lowest absolute air-leaving velocity, also putting as little work on the diffuser as possible. Straight vanes (*b*) are naturally the medium between the two extremes and their relative use will depend on the diffuser design. The simplicity of manufacture of straight vanes, compared to the curved types, also enters into consideration.

Following the line of discussion as used with axial compressors, the power input into a straight vaned impeller is given by

$$\frac{W\,u^2}{550\,g}$$

where u = peripheral velocity of impeller ft./sec., being proportional to the square of the speed as before.

The pressure ratio per stage is given by

$$\frac{P_2}{P_1} = \left[\frac{u^2}{g\ J\ C_p\ T_1} + 1\right]^{\frac{n}{n-1}}$$

for a straight-vaned impeller, both formulae ignoring all losses.

C_p = specific heat at constant pressure.
T_1 = absolute air temperature at inlet.
J = mechanical equivalent of heat (1400 ft. lbs. per CHU),

the other notations being the same as in the formulae quoted for axial compressors.

Results of tests on aero-engine superchargers are depicted in Fig. 22.

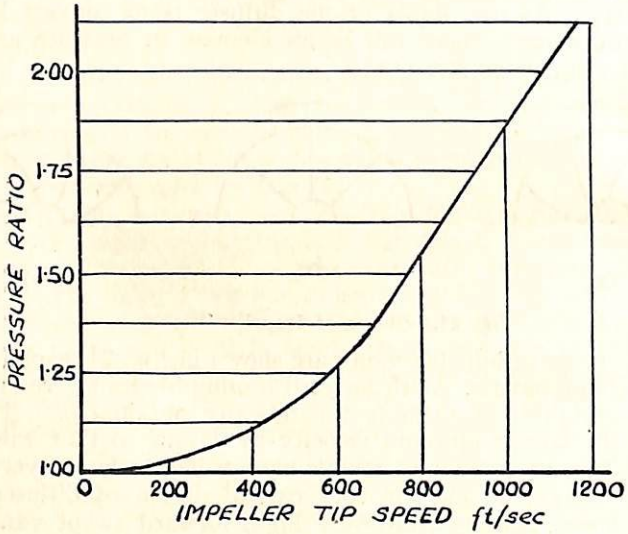

Fig. 22.*—Impeller Tip Speed and Pressure Ratio of Radial Compressor.

This shows that much higher pressure ratios are obtainable than with an axial compressor and, furthermore, that pressure ratio rises more rapidly with an increase in impeller speed, centrifugal compressors being essentially high-speed machines. Similarly to the previous description, we can talk here of adiabatic and mechanical efficiency.

As stated earlier, diffuser design is extremely important with the radial compressor and this will be discussed now. Air, when leaving the impeller, has both a radial and tangential component of velocity. The shape of diffuser vanes is such that the air is guided along the theoretically correct path for its design point, but as the speed varies, so does the angle at which the air leaves the impeller and the diffuser does not then work at the optimum conditions owing to rapid increase in breakaway. This is evident from the compressor characteristic. Diffusers with adjustable vanes have been incorporated in the compressor designed by Messrs. Brown-Boveri of Switzerland, but are still in the experimental stage.

* J. G. Keenan, "Gas Turbines and Jet Propulsion."

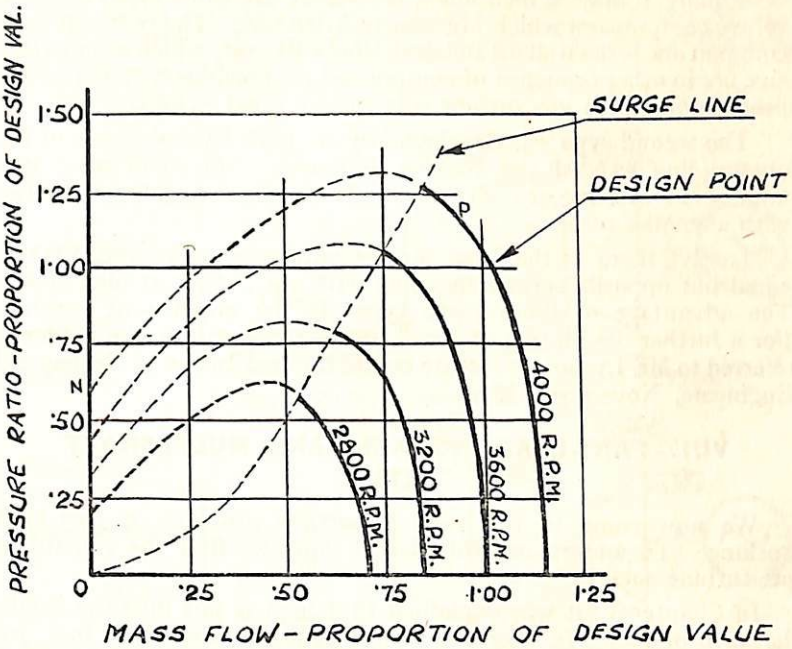

Fig. 23.—Pressure Ratio and Massflow of Radial Compressor.

The characteristic of a centrifugal compressor, Fig. 23, is very similar to its axial counterpart except that the curves are flatter than with the latter. The surge line is also indicated, the reason for its existence being the same as before, although owing to the flatness of the curve a centrifugal compressor is less liable to surge. Also with aircraft gas turbines the smaller frontal area of the axial compressor compared to the radial with equal massflow is an important consideration.

Summing up, it can be stated that the maximum efficiency of the centrifugal type is less than that of the axial. Also, as the stage pressure ratio of the centrifugal type is of the order of 4, it is mainly a high speed, low throughput machine. For higher pressure ratios, it would be easier to produce multi-stage axial compressors than centrifugals.

A combination of the two types is also in existence, this design trying to combine the advantages of both, usually several stages of axial compressor followed by one impeller. Axial compressors with adjustable blades have also been evolved. Although this practice may be established, the writer believes that axial and radial compressors will survive, both having their particular fields of application.

Finally, it may be mentioned that there are three types of spiral rotary compressors which are now in existence. The first and most common one is the well-established Roots Blower, which is in extensive use in other branches of compressed air machinery, but has not been employed in gas turbine sets to any great extent.

The second type was developed by A. J. R. Lysholm, one of the outstanding Swedish gas turbine designers. The main rotor is a large screw rotating at high speed and engaging for sealing purposes with a smaller rotor.

Lastly, there is the Imo screw-compressor, consisting of two equal but opposite screws engaging with each other at high speed. The advantage of these three types is the absence of surging. For a further description of spiral rotary compressors, the reader is referred to Mr. Lysholm's lecture before the Institution of Mechanical Engineers, November, 1943.

VIII.—PART-LOAD WORKING AND MULTI-SHAFT SETS.

We now come to the very important problem of part-load working. To understand this, let us consider first the control of gas turbine sets.

In Chapter V. it was explained that heat is put into the set in the form of fuel. If, therefore, we reduce the amount of fuel, we control the power output of the set. But let us look at the effect of this fuel reduction on the compressor and turbine.

The amount of air compressed is dependent on the compressor speed and remains constant with constant speed. If less fuel is injected into the combustion chamber, the inlet temperature to the turbine drops and so does its speed. Owing to the turbine being coupled to the compressor, the latter reduces speed and a drop in the compression ratio results.

Blade angles of both compressor and turbine are designed for full power output and at part-load therefore a drop in their respective efficiencies will result and also in the overall efficiency of the set. Furthermore, in the turbine there will be a change in the amount of work done per stage, the first stages doing more work and the last little or even negative work.

With power stations the requirements are constant speed and varying load. But as with constant speed the compressor delivers the same amount of air as at full load, whereas less fuel is supplied for part-load, the gas supplied is less than can be coped with by the turbine and the latter is therefore being "starved." The gas turbine set described so far is consequently unsuitable for constant speed, varying torque operation and for that reason multi-shaft sets are being increasingly employed.

In these sets the function is clearly divided into a charging section, *i.e.*, a turbine driving a compressor, and a power section, that is a turbine delivering external work. Fig. 24 (a) shows a two-shaft arrangement.

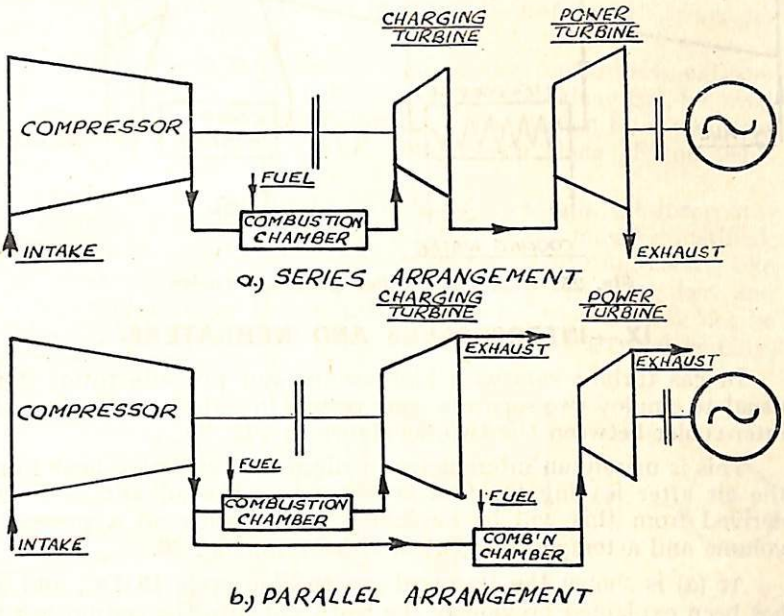

Fig. 24.—Two-Shaft Gas Turbine Sets.

The charging turbine is coupled to and therefore drives the compressor as in a simple set. But in this case the exhaust gases from the charging turbine pass to another low-pressure power turbine, which in turn drives the generator. The advantage with this set lies in its part-load working, allowing the speed of the charging set to drop to its required value at off-peak hours and keeping the power turbine running at the same speed but at reduced power.

A parallel arrangement is shown at (b), the difference being a separate combustion chamber for the two turbines and their independence from each other. Owing to the latter fact, this set is more advantageous to run at part-load conditions and is specially suitable for driving alternators and generators.

The above are only two examples of a variety of possible arrangements, each having its own characteristic and application.

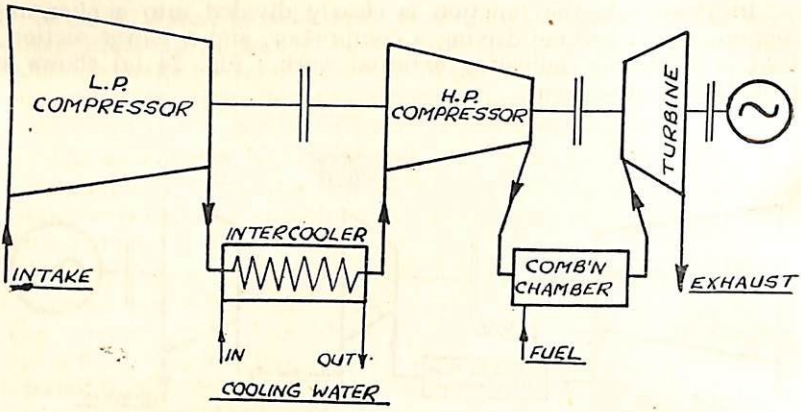

Fig. 25.—Gas Turbine Set with Intercooler.

IX.—INTERCOOLERS AND REHEATERS.

In gas turbine sets with high compressor pressure ratios it is usual to employ two separate compressors in series and to place an inter-cooler between the two, as shown in Fig. 25.

This is merely an interchange medium for extracting heat from the air after leaving the first compressor. The advantage to be derived from this will be explained by reference to a pressure-volume and a temperature-entropy diagram, Fig. 26.

At (a) is shown the standard gas turbine cycle BGDC, and as has been explained previously the work put into the compressor is represented by area ABGE. If now the compression line BG had been adiabatic, as it is under ideal conditions, it would follow BF

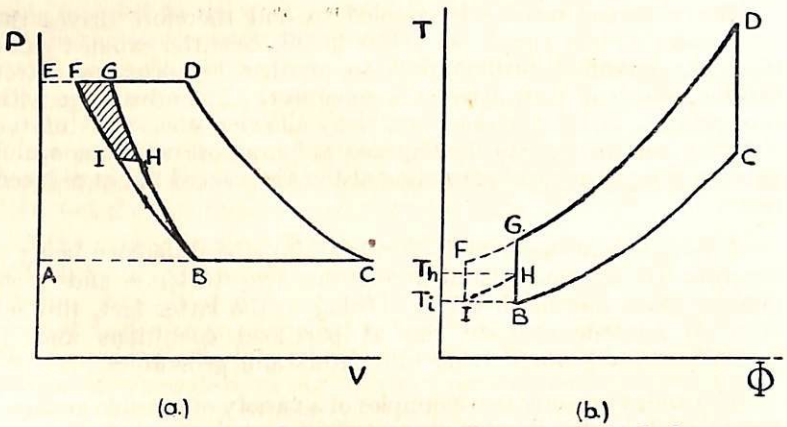

Fig. 26.—Effect of Intercooler on Thermodynamic Cycle.

and the useful external work of the set be increased by area BFG. Clearly, adiabatic compression is impossible, but inter-cooling does achieve it partly. Compression in the low pressure compressor follows the line BH, after which cooling at constant pressure takes place along HI until the adiabatic line is reached. Compression is then completed along IF, the increase in useful work being represented by area HIFG.

The same cycle notation is used in the temperature-entropy diagram, Fig. 26 (b). L.P. compression proceeds along BH, followed by constant pressure inter-cooling HI, accompanied by a temperature drop from T_h to T_i. H.P. compression along IF completes this part of the cycle.

If the overall compression ratio is large enough, the incorporation of three compressors and two inter-coolers may be justified, but it must be mentioned here that the use of an inter-cooler, like a heat exchanger, is invariably accompanied by pressure loss and if this is unduly large the inclusion of the inter-cooler may not be justified. Also their use is tied up with efficiencies and employment of heat exchangers.

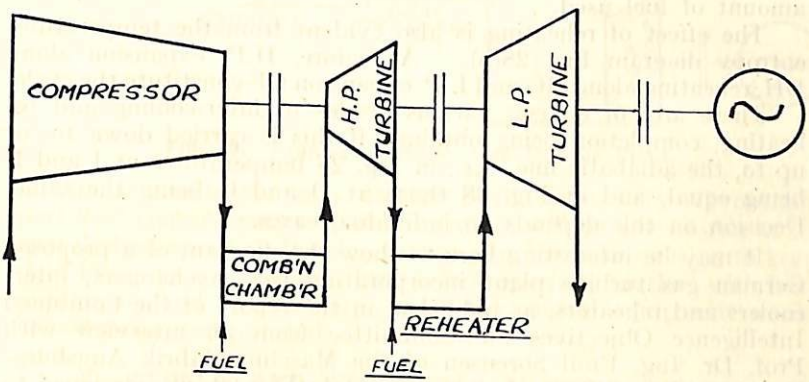

Fig. 27.—Gas Turbine Set with Reheater.

A further addition to a gas turbine set may take the form of a reheater, as shown in Fig. 27, although the gain in thermal efficiency is very small, unless a heat exchanger be employed also. Reheaters are in fact merely combustion chambers interposed between H.P. and L.P. turbines. The influence of reheating on the thermodynamic cycle is depicted in Fig. 28. BCDE is the simple gas turbine cycle with IDEA representing the work done by the turbine in diagram (a). If the reheat cycle be employed, the gases expand along DH in the H.P. turbine, are then reheated along HG, followed by expansion along GF in the L.P. turbine. Even with a heat exchanger the gain in thermal efficiency in a reheat cycle is only

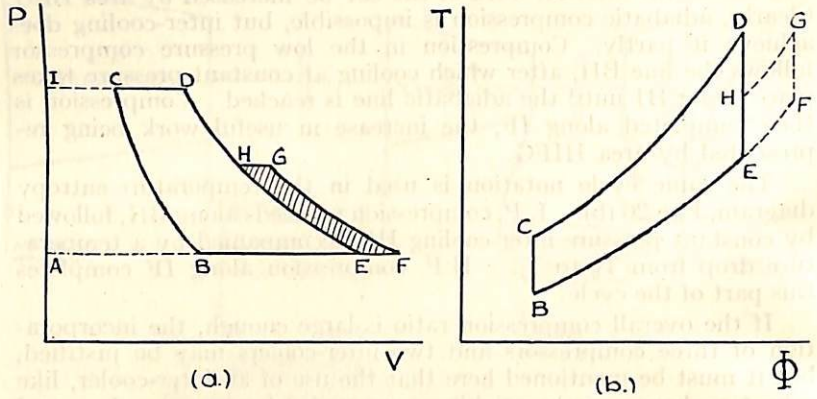

Fig. 28.—Effect of Reheating on Thermodynamic Cycles.

slight, but there is a marked improvement in power output per unit air consumption, accompanied, of course, by an increase in the amount of fuel used.

The effect of reheating is also evident from the temperature-entropy diagram Fig. 28(b). As before, H.P. expansion along DH, reheating along HG and L.P. expansion GF constitute the cycle.

There are, of course, various grades of inter-cooling and reheating, completion being obtained if this is carried down to, or up to, the adiabatic line, *i.e.*, in Fig. 26 temperatures at I and B being equal, and in Fig. 28 those at D and G being the same. Decision on this depends on individual cases.

It may be interesting here to show the diagram of a proposed German gas turbine plant incorporating heat exchangers, inter-coolers and reheaters, as published in the report of the Combined Intelligence Objectives Sub-Committee from an interview with Prof. Dr. Ing. Emil Sorensen of the Maschinenfabrik Augsburg-Nurnberg A.G. This plant, as shown in Fig. 29, was designed to develop 12,000 kW. at a thermal efficiency of 35% compared to about 18% of a simple cycle.

In this cycle, air is compressed in an L.P. compressor, passes through an inter-cooler and then to an intermediate pressure compressor, these two machines being coupled together. After a second cooling operation there follows H.P. compression and passage through a heat exchanger. The air is then heated in a combustion chamber and expands through a H.P. turbine, driving the two-coupled compressors mentioned previously. After reheating comes further expansion in an intermediate pressure turbine, driving the I.P. compressor. There is final reheating before passage through the power turbine (L.P.) and exhaust *via* the heat exchanger.

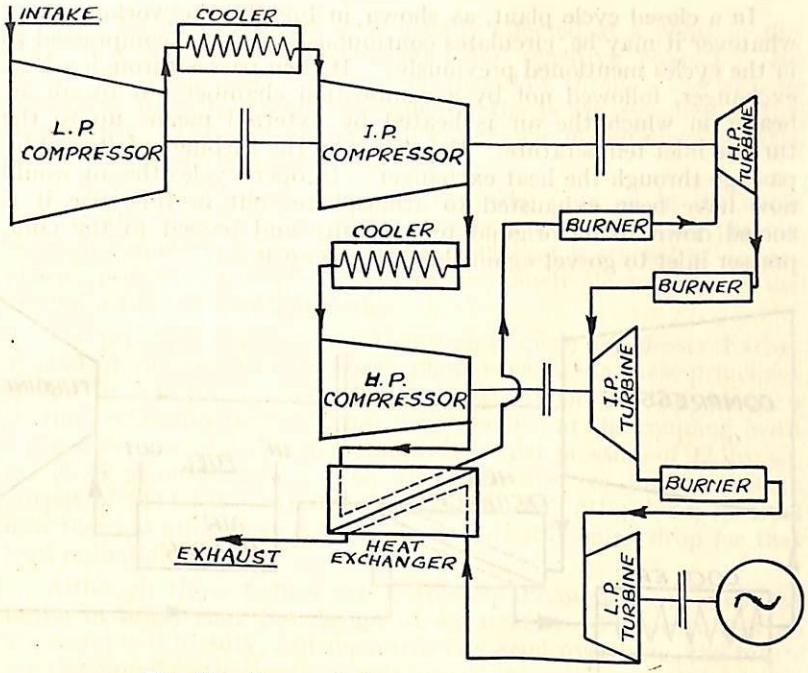

Fig. 29.—Proposed German Gas Turbine Plant.

An arrangement such as this will have very favourable part-load characteristics as each rotor set will be able to run at its own desired speed. It calls, though, for high-class design to keep duct losses to a minimum, these being especially marked with heat exchangers and inter-coolers. Also the control equipment would be somewhat complicated.

X.—CLOSED AND OTHER GAS TURBINE CYCLES.

All cycles discussed so far have used atmospheric air as a working medium, obtaining it from their surroundings and rejecting it again into this "limitless" reservoir. Naturally the reason for this is the universal availability of air, coupled with the fact that it can be obtained without cost, a not negligible advantage, although the air must be passed through a filter for purification. Air has to be accepted whatever its thermal qualities, and despite the fact that it may not be the ideal working medium for the gas turbine. A further "tag" on the use of air is the atmospheric pressure of approximately 14·7 lbs./sq. in. before compression. Also, as the power of a gas turbine set varies with the mass-flow, this can only be increased by greater inlet area, a very costly absorption of valuable space.

In a closed cycle plant, as shown in Fig. 30, the working fluid, whatever it may be, circulates continuously. Air is compressed as in the cycles mentioned previously. It then passes through a heat exchanger, followed not by a combustion chamber but by an air heater in which the air is heated by external means up to the turbine inlet temperature. Expansion in the turbine is followed by passage through the heat exchanger. In open cycles the air would now have been exhausted to atmosphere, but in this case it is cooled down to its original temperature and passed to the compressor inlet to go yet again through the cycle.

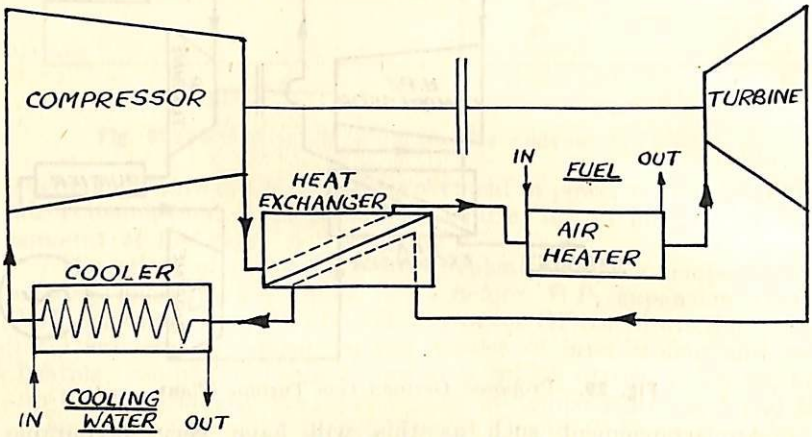

Fig. 30.—Closed Gas Turbine Cycle.

The principal advantage of this cycle lies in the fact that mass flow can be increased without increasing the size of the set. This is done simply by raising the general pressure level of the set, *i.e.*, starting at the compressor inlet not with one atmosphere but with, say, five atmospheres. The efficiency, therefore, only depends on the pressure ratio and not on the absolute value of pressures. Furthermore, compressors can be designed to work at their peak efficiency all the time and no allowance has to be made for part-load working by a compromise with blade inlet and outlet angles.

This high pressure circulation affords easy control of the set and practically constant part-load efficiency. Load is reduced by opening a valve in the circuit, thereby lowering the general pressure level and the massflow and power output. Air velocities remain practically constant and so do compressor and turbine efficiencies. To increase power again, pressure throughout is increased, a high pressure cylinder being therefore an essential component of a closed cycle plant.

Blow off and pressurising is usually effected at a point in the cycle just before the turbine inlet for sensitive control. The reason for this is that a change in massflow through the turbine will affect its output and consequently the power supplied to the compressor. This in turn will cause a change in compression ratio and will result in rapid load change.

A further advantage is gained by using pure air in the cycle and by the absence of corrosive products of combustion passing through turbine and heat exchanger. The set is started at atmospheric pressure and the pressure is slowly increased to the desired value as speed increases.

The principal exponents of this type of cycle are Messrs. Escher Wyss Ltd. of Zurich and a power plant working on these principles is now in operation. According to a test report on that plant, a thermal efficiency of 32·6% has been reached at the coupling, with a pressure ratio of 3·77 and a compressor inlet pressure of 92 lbs./sq. in. (6·25 atmospheres). The above figures are for a coupling output of 2111 kW. Another test at 446 kW. (21% load) resulted in a thermal efficiency of 25·5%, a remarkably small drop for that load reduction.

Although those figures are extremely favourable, it must be borne in mind that the design of air heaters still constitutes an appreciable difficulty, but once this has been overcome, the future for the closed cycle seems promising.

In this country the English Electric Co. has proposed a closed turbine cycle which uses a monatomic gas instead of air as a working fluid. The advantage claimed for the use of this is the ratio of 1·66 of the specific heat at constant pressure to the specific heat at constant volume (this ratio is 1·4 for air). This results in a lowering of the pressure ratio required for compressor and turbine to give maximum efficiency at chosen temperature limits. Furthermore, with an increase in the gas density there is an increase in mass-flow resulting either in smaller turbine dimension or greater power with existing dimensions. Also the use of inert gases will cause an appreciable reduction of corrosive action on blades and ducts.

In the United States of America an interesting cycle, which can best be described as semi-closed, is under development by the Westinghouse Company and a diagram appears in Fig. 31. It consists of a closed cycle part (shown heavy) and an open cycle part for circuit charging (shown faint), sharing a common combustion chamber. The closed cycle part is as described previously, complete with heat exchanger and water cooler. But in this case the gases from the combustion chamber pass through an auxiliary turbine driving an auxiliary compressor for supercharging purposes. The disadvantage with this arrangement is the presence of hot and corrosive gases in the main cycle and separators must be incor-

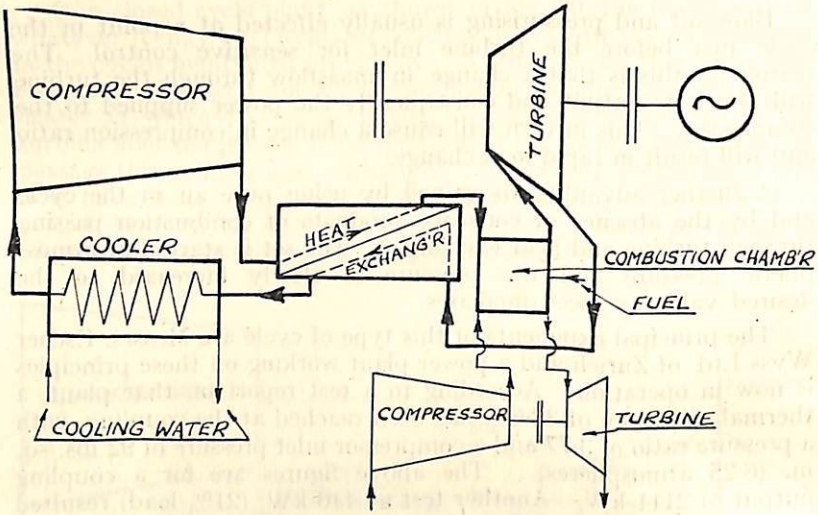

Fig. 31.—Westinghouse Gas Turbine Cycle.

porated to remove all solid matter of combustion before entry to the turbine.

It may be possible to evolve a cycle similar to the ordinary closed cycle where the air and fuel used for the air heater form part of an open cycle, so that power can be taken off the closed and open cycle parts.

There have also been schemes employing such working media as alcohol, carbon tetrachloride, sulphur dioxide, mercury vapour, etc., but in the writer's opinion these have not yet proved themselves successful. In fact, they may not even have passed the design stage and therefore are only mentioned here in passing.

XI.—PRESENT-DAY APPLICATION AND FUTURE TRENDS.

Having so far described in broad outline the problems confronting the designer of gas turbines to-day, we may well ask ourselves how this new prime-mover is being fitted into various branches of industry, whether the application of gas turbines is proving a promising proposition compared to other rivals and what are the prospects and future trends of development.

There is no doubt that in the aircraft industry gas turbines in the shape of pure jet propulsion or gas turbine propeller driven aircraft have shown their value, but it cannot be said that they have already displaced their I.C. reciprocating predecessors, the design of which has been brought to a truly remarkable pitch.

Nevertheless, owing to the extensive development of this engine in the aircraft industry, a lot of valuable experience has been gained.

Traction engines, particularly in the field of railway locomotives, seem to have a strange fascination for all designers of prime movers and as was mentioned in one of the earlier pages a gas turbine locomotive, designed by Brown Boveri and Co. for the Swiss Federal Railways (see Bibliography) has been in operation since 1941, and has also been extensively tested. It may, therefore, be worth while to study this design more closely and compare it with other railway locomotives in use to-day.

A simple gas turbine cycle is employed incorporating a heat exchanger, but no separate power turbine. The cycle is in effect as shown in Fig. 12. The turbine drives a generator, supplying four D.C. traction motors, continuous output at the coupling being 2200 H.P. At 2000 H.P. output, the thermal efficiency is 17·4% and the inlet temperature 570°C., both of these latter figures having been increased in the design stage of more modern sets. Efficiency drops rapidly at part-load, being only just above 14% at 1000 H.P. Compared with ordinary steam locomotives, it may be noted that even with thermal efficiencies as obtained on this first Swiss design, it can compete with the 8 - 13% of the former. The length of the Swiss locomotive is less than steam or diesel engines of the same power, a not insignificant point in locomotive design.

Owing to the purely rotary motion, there are no unbalanced forces on the frame and track wear is very much reduced. The torque is uniform and although fuel consumption is higher than with diesels, the gas turbine can use a cheaper range of fuels.

The interest of locomotive engineers in gas turbines has grown recently and several American firms have project designs of such engines in their drawing offices. In this country, the Great Western Railway has two gas turbine locomotives on order, one being again built by Brown Boveri Co. with the experience gained by their first design. The second one is in the hands of Messrs. Metropolitan Vickers Ltd. Both these engines employ electric transmission, but there is no reason to believe that mechanical transmission should not be used, *i.e.*, torque transmitted *via* a gear box directly to the driving axle.

Summing up the case for the gas turbine locomotive, it can be said that if this problem is carefully handled there should be no reason why many of our main-line engines of the future should not employ this prime mover, even with mechanical transmission.

One of the earliest applications of gas turbines to industry is the one built by Messrs. Brown Boveri Co. for the Sun Oil Company of Philadelphia, U.S.A., for use in the Houdry oil refinery process, which requires the supply of large quantities of compressed air. The first unit has been in operation since December, 1936, with

remarkably good results and there are at present 17 such plants in operation in the U.S.A.

Gas turbines in connection with electric generating plant have been mentioned, referring to the Escher Wyss closed cycle. But since early in 1940, a 4000 kW. gas turbine emergency plant employing the open cycle without heat exchanger has been doing standby duty in Neuchatel, Switzerland. Thermal efficiencies of 18% were obtained with turbine inlet temperatures of only 550°C.

The largest gas turbine plant to date was under construction in 1945 by Brown Boveri Co. It was designed to develop 10,000 kW. and will be installed eventually in the Bucharest main power station (Rumania). In view of the size of the plant and the recent date of its design, some representative figures may be of interest. There is no heat exchanger ; owing to the size, the increased cost did not warrant it. Temperature at turbine inlet is 600°C. and thermal efficiency 21·5%. The air swallowed reaches the amazing figure of 295 tons per hour and the cooling water required is 66,000 galls. per hour. The set reaches full speed ten minutes after starting.

Velox boilers using gas turbines to supply compressed air for boiler firing have been in use for many years and are, if one may use the phrase, almost commonplace. They are outstanding for the quickness with which they can be started, the small space required and the high overall efficiency.

Gas turbines applied to marine installations have been of rather recent innovation, in fact the first boat so propelled in this country was a motor torpedo boat of the Royal Navy, tested in August, 1947. The installation was built by Metropolitan Vickers Co. Also, the B.T.H. Company has under construction a marine gas turbine plant to be used on an oil tanker. Both these plants incorporate heat exchangers.

In the U.S.A. a gas turbine of 3500 H.P. rating has been constructed by the Allis Chalmers Mfg. Co. and was tested in 1948. The cycle is similar to the one shown in Fig. 24b, plus a heat exchanger. The first published tests yielded thermal efficiencies of the order of 25%, but higher figures have been reached with later sets, which employ inter-coolers and series turbines as Fig. 24a, together with reheaters.

The American firm of Elliott has built a 2500 H.P. turbine which is claimed to have an efficiency similar to a steam plant. This design uses the Lysholm rotary compressor and it is stated that the efficiency can be raised to 34%, the plant embodying three compressors, two inter-coolers, three combustion chambers and a heat exchanger. Present maximum turbine temperature is 700°C.

So far the gas turbine sets mentioned had their output quoted in thousands of horse-powers and it is therefore interesting to note that the American Westinghouse Co. has developed a set driving an electric generator for aircraft auxiliaries developing 24 kW.

Finally, a few words about the use of pulverised coal as fuel for gas turbines. No startling results have yet been achieved with this novel idea, but development is under way and useful results will no doubt be obtained.

In this pamphlet the writer has attempted to describe the problems involved in the design of a modern gas turbine set and the various cycles and arrangements that can be adopted. Too much development work is still going on in various countries for definite conclusions to be reached as to where the gas turbine will eventually establish itself and in what form.

At the risk of being contradicted, the writer would like to state his belief that, depending on the work of metallurgists, turbine inlet temperatures of 900°C. will be in the grasp of future engineers ; heat exchangers, where efficiencies of 75% are being achieved to-day, will, by patient development, be pushed still higher in output, and turbine and compressor efficiencies may well be in the nineties. With these component efficiencies the thermal efficiency of the set will no doubt reach 40% and will then be a serious competitor to the modern highly developed diesel engine.

The bibliography at the end of this pamphlet represents the kernel of published information. German works are also in existence but are not included as the writer was unable to obtain these owing to wholesale destruction of stocks. There are references to current developments in the gas turbine field in the technical journals of Britain, Switzerland and U.S.A., and the U.S.S.R. also is very much alive to the future of gas turbines.

Most of the books mentioned in the list give information of a similar character to this pamphlet, but in a much more complete fashion. Nevertheless a work on gas turbines, similar in thoroughness and covering such a vast field as A. Stodola's volumes on steam turbines has, in the writer's opinion yet to be published, and it seems the present time is too early for such information to appear. Nevertheless the gas turbine has made its entry and will increase in importance as more and more data are collected from its various applications.

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